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INTERNAL COMBUSTION ENGINE

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No. 501

EFFECT OF INTAKE PIPE ON THE VOLUMETRIC EFFICIENCY
OF AN INTERNAL COMBUSTION ENGINE

By Antonio Capetti

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TECHNICAL MEMORANDUM NO. 501

EFFECT OF INTAKE PIPE ON THE VOLUMETRIC EFFICIENCY
OF AN INTERNAL COMBUSTION ENGINE.*

By Antonio Capetti.

Summary

The writer discusses the phenomena of expansion and compression which alternately take place in the cylinders of four-stroke engines during the induction process at a high mean piston speed due to the inertia and elasticity of the mixture in the intake pipe.

All engine builders usually take care of this problem by retarding the closing of the inlet valve. They know, however, that the length of the intake pipe affects the maximum power of the engine through its effect on the carburetion.

In order to determine the effect on the charging of the cylinder, the writer had previously worked out a method of theoretical research and had arrived at some interesting conclusions on the existence of a most favorable condition for the charging, which should take place in accordance with a certain ratio of the pipe length to the piston stroke.

The present paper is intended to demonstrate theoretically the existence of a most favorable pipe length for charging,

*"L'influenza della tubazione aspirante sul rendimento volumetrico dei motori a combustione interna." From Annali della R. Scuola d'Ingegneria di Padova, December, 1927, pp. 393-404.

which can be adopted without disturbing the carburetion by using an additional pipe preceding the carburetor. The effect of using the same pipe for several cylinders is also considered.

Reference is made to the experimental confirmation this theory has received both from American experiments (by the National Advisory Committee for Aeronautics) with a one-cylinder engine and from the writer's experiments with a four-cylinder engine, which are more fully described in the appendix.

The writer concludes with an analysis of the possible effects of special devices for inducing or damping the inertia, such as "capacities" in series or in parallel.

I. Premises

The intake pipe of an internal combustion engine with a four-stroke cycle is traversed by a pulsating current of air or carbureted mixture.* It is obvious, therefore, that all the phenomena of periodically varying currents, due to the inertia and elasticity of the fluid, may be manifested.

These phenomena can be studied individually by considering all the contributing causes, namely, the ratio between the cross-sectional areas of the pipe and cylinder; the obliquity of the connecting rod; temperature and pressure differences between the residues in the cylinder and the fresh charge; the friction-

*In the rest of this article I shall use the term "air," since the very small quantities of carburetant or fuel present have no appreciable effect on the phenomenon under consideration.

al resistance of the variable valve opening, intake pipe and carburetor.*

These phenomena may also be considered theoretically by comparison with other analogous phenomena, such as alternating electric currents, notwithstanding some substantial differences from our case, which involves only simple pulsating currents. The inertia which the air opposes to the acceleration imparted to it by the piston is not necessarily harmful to the charging of the cylinder, though it reduces the motion of the air with respect to the motion of the piston. At the first stroke, therefore, the pressure in the pipe is reduced and the cylinder is only partially charged (Fig. 1). This condition lasts until the suction is not transmitted to the outside, i.e., until the front of the expansion wave has not reached the mouth of the intake pipe. From this moment, however, a compression wave begins to penetrate the intake pipe and push the air into the cylinder, even after the piston has reached the lower dead center and begun its return stroke.

It is obvious, therefore, that the initial retardation in charging the cylinder may be offset by the succeeding inrush of air. If the phase displacement reaches a specially favorable value, a real supercharging should be effected by inertia, since

*This particular investigation has been already described in two articles: "Contributo allo studio del flusso nei cilindri dei motori veloci" (Ingegneria, 1923) and "Fenomeni dovuti all'inertzia nella alimentazione dei motori aeronautici" (Annali della R. Scuola di Ingegneria di Padova, 1927).

the cylinder may remain charged with compressed air throughout the whole or almost the whole of its length, in which case the contained air will have a density and pressure somewhat greater than the surrounding atmosphere.

II. Best Length of Intake Pipe

The best length of the intake pipe and consequently the maximum volumetric efficiency of the cylinder can be easily determined approximately by the following elementary reasoning.

The best dephasement is that at which the wave of maximum compression reaches the cylinder at the instant the latter has its maximum volume, i.e., when the piston is at its lower dead center. The wave of maximum compression is produced, however, by the refraction of the wave of maximum expansion or, in other words, by the reaction to the maximum reduction of pressure, which leaves the cylinder at about the middle of the piston stroke, because the piston is then moving at its maximum speed. It is therefore necessary for the time required for the wave to traverse the intake pipe twice (once in each direction) to equal the time required for the piston to make about half a stroke.

Then, if w indicates the mean velocity of propagation of the waves (absolute velocity of sound) expressed in meters per second, and if v represents the mean piston speed, likewise in meters per second, the two time intervals to be equated are

$2l/w$ and $c/2v$. Hence the best ratio of the pipe length to the piston stroke is

$$\frac{l}{c} = \frac{w}{4v}$$

We may give a different form to the ratio just found by introducing the number of revolutions per minute n , which is related to the mean piston speed v and to the stroke c by the simple formula $v = \frac{n c}{30}$, and we obtain

$$l = 7.5 \frac{w}{n} *$$

Assuming $w = 340$ (at a temperature of 15°C), we obtain the values given in Table I.

Table I.

n	1000	1500	2000	2500	3000	3500	4000	4500	5000
l	2.55	1.70	1.28	1.02	0.85	0.73	0.64	0.57	0.51

These values are somewhat greater than those obtained by careful calculation and experimentation. They may be corrected by observing that the velocity of propagation of the waves is not simply that of sound in a quiet medium, it being necessary to add or subtract the velocity of the air itself in the direction of propagation. Letting u represent the velocity of the air,

*It is interesting to note that this formula expresses the "resonance" between the motion of the piston and the vibrations in the pipe. The period of free vibration of a tube open at one end is $4l/w$. The intake period of the piston is $30/n$. Equation (2) affirms that these two periods are equal. Thus the phase displacement is of one-half stroke, or 90° , as in resonance phenomena.

the first of the two above-mentioned time intervals will then be

$$l \left(\frac{1}{w+u} + \frac{1}{w-u} \right) = \frac{2l}{w} \frac{w^2}{w^2 - u^2} \quad (1)$$

and equation (1) becomes

$$\frac{l}{c} = \frac{w}{4u} \left(1 - \frac{u^2}{w^2} \right).$$

The second line of Table II gives the values of the best pipe-stroke ratio as calculated by equation (1), while the third line gives the corresponding values according to equation (3).

For the velocity u , we can use twice the mean value, since the waves of maximum expansion and compression are being propagated while the velocity in the intake pipe is varying in the vicinity of its maximum. We must, of course, adopt as the mean value, the same v multiplied by the ratio of the cylinder section to the pipe section. In calculating Table II this ratio was assumed to be 5.

Table II.

n	4	6	8	10	12	14	16	18
$\frac{l}{c}$	21.2	14.2	10.6	8.5	7.1	6.1	5.3	4.7
$\frac{l}{c}$	20.9	13.8	10.0	7.8	6.2	5.0	4.0	3.4

III. Results of More Complete Theoretical Analyses

We have already theoretically illustrated a special function of the intake pipe as an active intervening element in charging a cylinder. The preceding considerations are incomplete, however, in various respects. As already mentioned, their very nature does not permit the investigation of the accessory causes governing the phenomenon.

For this purpose, we need only to repeat the conclusions reached in the above-mentioned investigations. Figure 2 shows the theoretical curves of variation in the volumetric efficiency and also the curves for the theoretically most favorable angles of lag in the closing of the intake valves, as plotted against the ratio of the pipe length to the piston stroke. In plotting these curves, no account was taken of the effect of the residues in the dead spaces in the cylinders.

Another investigation demonstrated that the residues, in so far as they have a pressure higher than the atmospheric pressure, lower the volumetric efficiency considerably without affecting the best lengths. On the contrary, these lengths are shortened in so far as the temperature of the residues is higher than that of the atmosphere.

IV. Resistance of Intake Pipes

Two further important comments should be made regarding the reasoning in Section II. The first one concerns a different aspect of the functions of the intake pipe, namely, its frictional resistance to the passage of the air. From this viewpoint the best form of pipe would be the shortest and largest possible.

The combined effect of the two functions, active and passive, is therefore a reduction in the inertia and in the best length by an amount which increases with the velocity of the air and therefore with the piston speed. The existence of a best length is not eliminated, however, as has been demonstrated by experimentation and calculation.

V. Effect of the Number of Cylinders

The second comment on the reasoning in Section II concerns the adaptation of the same intake pipe to several cylinders of the same engine. Thus far we have assumed that, when the intake valve opens, the air in the pipe has the same pressure as the surrounding air. This hypothesis seems to be satisfactory so long as the pipe supplies only one cylinder. In this case the cylinder remains inactive for about three piston strokes. During this time the waves left in the pipe at the end of the suction stroke traverse it in both directions a great number of times; 12 times, for example, if it concerns the best length according to formula (1) or (2). The frictional resistance al-

most completely damps these oscillations.

If two, three, four, or even six cylinders, as in several modern small automobile engines, are supplied by the pipe or by the larger part of it, the damping action does not have time to take effect and the phenomenon of inertia is altered. Due to the variety and complexity of the possible cases, it is not often possible to predict just how things will happen. It may be assumed, however, that the supercharging effect of the inertia is diminished by the multiplicity of the suction.

In fact, it is obvious that the best cylinder charging occurs in the vicinity of the lower dead center of the piston on the arrival of a wave of maximum compression following a wave of maximum reduction in pressure. If, for example, we have four cylinders, the suction begins in one cylinder just as it ceases in another, and therefore the wave of compression generated in the latter cylinder retards the pressure reduction in the former cylinder with all its consequences.

By way of example, we will investigate in greater detail this very common case of a four-cylinder, four-stroke-cycle engine for a considerable number of successive strokes by the analytic graphic method already mentioned. Figure 3 shows the pressure-volume curves for the first four intake phases of the cylinders. Figure 4 shows the corresponding curves for the four succeeding phases. It is obvious that we are approaching a pressure region quite different from that of the single cylin-

der (dash curve in Fig. 4). The pressure reduction moves toward the lower dead center, while the successive recoveries grow smaller.*

Table III gives the values found for the volumetric efficiency of the successive intake strokes in the different cylinders, all these values being much smaller than the corresponding values for the single cylinder or (which amounts to the same thing) for the multiple-cylinder engine with separate intake pipes.

Table III.

Theoretical Volumetric Efficiency of a Four-Cylinder Engine

$$1/c = 6 \quad v = 12 \quad \text{lag} = 38^\circ$$

1.37	1.20	1.08	1.04	1.14	1.06	1.08	1.14	1.08	1.11
Separate pipes	Cyl. 1 Cycle I	Cyl. 2 Cycle I	Cyl. 3 Cycle I	Cyl. 4 Cycle I	Cyl. 1 Cycle II	Cyl. 2 Cycle II	Cyl. 3 Cycle II	Cyl. 4 Cycle II	Mean

It can be easily seen from the figures, and still better from the table, that the rate of pulsation which tends to become stable has a different period from that of the engine, the period being triple in this case. This allows us to assume that the best length may change in passing from the single to the multiple charging.

*The dotted lines in Figs. 3 and 4 are tangent to or secant through the end of the adiabatic portion of each induction-pressure curve. Since, however, by hypothesis, these portions of the curves are adiabatic at varying pressure, the above-mentioned lines show whether the lag in the closing of the inlet valve is too small, just right, or too large.

VI. Experimental Verification

The theoretical predictions on the behavior of the intake pipes have been satisfactorily confirmed by the few verifying experiments thus far made. We will here refer to only two series of tests, the first series, comprising the tests made by the United States National Advisory Committee for Aeronautics.*

The tests were made with a single-cylinder (Liberty) engine (of 5-inch bore and 7-inch stroke) which was operated by an electric motor during the tests. The test conditions differed, therefore, from those in ordinary practice, although they approximated the conditions in our first experiments, which yielded the curves of Figure 2.

The results of the American tests are shown in Figure 5, the abscissas being, as in Figure 2, the ratios of the length of the intake pipe to the piston stroke, while the ordinates, in the absence of other indications, are the ratios of the final compression pressures, measured at various velocities, to the compression pressure at the minimum velocity. They are, therefore, like the ordinates of Figure 2, representative values of the amount of the charge.

Comparison of the two figures shows a sufficient agreement of theory and practice for single-cylinder engines, despite the fact that the lag in the closing of the intake valve remained

*Robertson Matthews and Arthur W. Gardiner, "Increasing the Compression Pressure in an Engine by Using a Long Intake Pipe." N.A.C.A. Technical Note No. 180, 1924.

unaltered during the tests although, according to theory, this lag should have varied both with the velocity and with the length of the pipe.

The second series of tests was executed by the same parties with a four-cylinder engine, provided with a common intake pipe throughout the whole length, excepting for the short passages leading through the cylinder block to the different valves. The details of these tests are given in a brief appendix to this communication. We will simply state here that the engine was run continuously at full throttle and that careful control was maintained of all accessory conditions, such as the temperature of the cooling water, the ignition timing, the proportions of the carbureted mixture and the torque developed.

The test results, which the rather antiquated type of engine did not permit carrying to very high piston speeds, are shown in Figure 6, whose coordinates are the same as those of Figure 2, and in Figure 7, whose abscissas are the mean piston speeds.

Figure 6 shows that even in a four-cylinder engine with a single intake pipe, there is a practical advantage in increasing the length of the intake pipe up to a certain value. To be exact, we found that by using a pipe length of 0.85, 1.1, 1.2, or 1.5 m (i.e., 6, 8, 8.6, or 11 times the piston stroke) for corresponding mean piston speeds of 7, 6, 5, and 4 m/s, we obtained respective increases of 8.5, 10.5, 19.5, and 24% in the volumet-

ric efficiency. The phenomenon of supercharging by inertia was thus verified in its essential features even in this case.

The principal divergencies from the theoretical prediction for the single-cylinder engine (Fig. 2), apart from the easily predictable reduction in the absolute values of the volumetric efficiency, consist in the considerable reduction in the best lengths and still more (Fig. 7) in the reduction in the velocity of maximum charging for each length. Especially with regard to the latter divergence, a considerable effect may be attributed to the constancy of the lag in the closing of the intake valves. This was not verified, however, by the American experiments (Fig. 5), where the lag was also kept constant.

We are therefore led to conclude that the greater effect is due to the number of cylinders supplied by the same pipe. To this we may also attribute the peculiar shape of curve VII in Figure 7, namely, the presence of two speeds of maximum charging when the pipe is very long.

Figure 8 shows the "characteristic" curves (i.e., the M.E.P. plotted against the R.P.M.) of an engine functioning with various additional lengths of pipe, for the purpose of showing how far the greater charging of the cylinder, due to inertia in the long pipes, is converted into greater effective brake horsepower. These curves have even greater practical significance than those of Figure 7, in that they are not affected by the subjective criteria of measurement and evaluation

which may be involved in the determination of the volumetric efficiency (See the appendix). Of course care had to be exercised during the tests to keep nearly constant the other conditions affecting the mean effective pressure, namely, the composition of the mixture, the timing of the ignition, and the temperature of the water for cooling the cylinders.

In its general appearance, Figure 8 resembles Figures 7 and 6. They all show that the engine power (curves II, III, IV and V in comparison with curve I) is augmented by increasing the length of the pipe up to a certain limit (curve VI corresponding to 22 times the piston stroke). Even very long pipes have shown advantages, however, for crank-shaft speeds above 1300 R.P.M. This fact, already obvious from the experimental results represented by curves VI and VII, shows their paradoxical character with respect to the usual method of considering the effect of intake pipes, because it is at high speeds that we would naturally fear excessive resistances from very long pipes.

Lastly, we note that the maximum advantage found for the mean effective pressure, and hence for the engine power, is about 11%. It is, therefore, less than that found for the volumetric efficiency. This proves that the supercharging due to inertia is not gratuitous, but is obtained, as might be expected, at the expense of the engine torque.

VII. Effect of Air "Capacities" in Series or in Parallel

Having theoretically determined and experimentally confirmed the source of the beneficial influence exerted by the inertia of the air flowing through the pipes, it is natural for us to seek means to increase this inertia, which we have called an active function of the intake pipe, without lengthening the pipe, which simultaneously increases the frictional resistance.

There are many means to which we might resort. Among the simplest of these, in the case of a carburetor supplying several cylinders, we may mention the possible advantages obtainable by the choice of suitable points of connection for the branch pipes leading to the different cylinders, while among the more complex means, we may mention the introduction of special automatic or mechanically controlled valves for varying artificially the length of the active portion of the pipe.*

By way of illustration, we will here refer to only two particular cases pertaining to the experiments described in Section VI and in the appendix of this communication. We propose to consider whether and how we can vary the velocity of the air in the intake pipes of the engine by introducing a pipe section of large diameter (air capacity in series) or a tank connection

*Application has been made for patents for all the devices capable of increasing the inertia of the air for the purpose of supercharging, including the additional pipe.

(air capacity in parallel). If the capacity is inserted in the pipe near the entrance, its effect on the volumetric efficiency of the engine is almost zero. In fact, when the waves of expansion or reduced pressure, in retraversing the pipe, reach the greatly enlarged section of the capacity, they are refracted and form return waves of compression of little less strength than those generated by an external opening. Upon their arrival at the cylinder, the beneficial compression waves are therefore only slightly weakened during the first phase. In the second phase, however, the waves which have advanced beyond the enlarged section and reentered the capacity are, in their turn, refracted at the external opening of the latter and form waves of slight compression which facilitate the charging of the cylinder. Some advantage may result from these two phenomena if the length of the inserted capacity is suitable.

Exact calculation confirms all this and gives, for the case in which $v = 12$ m/s, $v/c = 6$, and for an air capacity with a cross section ten times greater than that of the pipe, a drop in the volumetric efficiency ranging from a loss of 3% (for capacities over 14 times the length of the piston stroke) to a slight advantage for capacities only 2-3 times the piston stroke.

As to the velocity in front of, and particularly at the external opening, it is not even greatly affected by the presence of the inserted capacity. In particular, there is no

benefit of equalization of the velocities necessary, for example, for making important measurements by manometric methods.

This is evident if we remember that the velocity in the connecting section between the capacity and the intake pipe is given with close approximation* by that of the adjacent pipe divided by the ratio of their cross-sectional area. Since the second velocity is only slightly altered, the same may be said of the first.

Moreover, this is demonstrated by the preceding calculations. It is only necessary to compare the curves a and b in Figure 9,** where the ordinates differ, but the general course, from zero to zero, is similar.

Let us now pass to the case of a capacity "in parallel," also located near the entrance. The effect on the charging of the engine is very small in this case also, because the wave finds a section so greatly enlarged at the point where the tank is connected that it is refracted almost the same as at an external opening. However, the effect, although small, is now of the opposite sign to that of the preceding case. In fact, while we first saw that, in a second period, the air tank was retraversed by compression waves (generated by the refraction of expansion waves at its open end), the tank is now retraversed by

*For the exact valuation, it would be necessary to take into account the ratio of the densities. This ratio, however, hardly reaches 1.05 for velocities of 100 m/s (328 ft./sec.) in the pipe at a temperature of 25°C.

**In order to make the figure clearer, the ordinates of the curve b are the velocities at the opening multiplied by 10, the ratio of the cross section of the air tank to that of the pipe.

expansion waves generated by the reflection at its closed end of similar waves coming from the cylinder. Lastly, the effect of the capacity on the velocity of the air flow into the pipe is very marked. It is only necessary to observe the curves c and d in Figure 9 which, respectively, represent the calculations we have made for the first intake stroke (or the first cylinder) and for the third stroke (or the third cylinder).*

*The solutions of the cases of wave refraction given in our previous papers are not sufficient for the analytical treatment of "capacity in parallel." The method of reasoning employed in Section 2b of the paper "Contributo allo studio del flusso" (Contribution to the Study of Flow) is easily applicable, however, to this case. It is only necessary to convert the first of the equations (6) into the form

$$u' \rho' \varphi + u \rho \varphi_0 = u'' \rho'',$$

in order to allow for the effect of the capacity in parallel and then to add the three equations for the waves propagated in it:

$$u^2 - u''^2 = \frac{2 k p}{k-1} \frac{p}{\rho} \left[\left(\frac{\rho''}{\rho} \right)^{k-1} - 1 \right]; \quad r - s = u; \quad r + s = \frac{2}{k-1} \sqrt{k} \frac{p}{\rho}$$

The solution of the system of nine equations then proceeds in a similar manner with the use of the auxiliary unknown quantities (two in this case, $\psi = \rho'/\rho''$ and $\psi_0 = \rho/\rho''$) and leads to the series of three equations solvable similarly to the equations (7).

$$u'' = 2 \frac{r \varphi_0 \psi_0 - s'' (\varphi \psi^{1.2} + \varphi_0 \psi_0^{1.2}) + r' \varphi \psi}{1 + \varphi \psi^{1.2} + \varphi_0 \psi_0^{1.2}}$$

$$u' = 2 \frac{r' (1 + \varphi_0 \psi_0^{1.2}) - r \varphi_0 \psi_0 \psi^{1.2} - s'' \psi_0^{0.2}}{1 + \varphi \psi^{1.2} + \varphi_0 \psi_0^{1.2}}$$

and likewise for u by changing φ, ψ, r' and r into φ_0, ψ_0, r and r' respectively, (Fig. 10). The values of ψ and ψ_0 are then found by trial, as explained in Section 2b of the above-mentioned paper.

A p p e n d i x

Experimental Determination of the Volumetric Efficiency
of an Engine Provided with Intake Pipes of Various Lengths

The engine used in these experiments was a four-stroke-cycle Fiat 53B with four cylinders of 8 cm (3.15 in.) bore and 14 cm (5.51 in.) stroke; compression ratio 4.2; lateral valves with about 60° lag in the closing of the intake valve.

The intake passages through the cylinder block to the single carburetor have a mean length of about 40 cm (15.75 in.) between the intake valve of the cylinder and the mouth of the carburetor. Consequently, this was the minimum length used in the tests. In order to increase this, we added a pipe of 35 mm (1.38 in.) diameter (hence, with a cross section equal to $1/5$ of that of the cylinder) and of variable length up to about 4 m (13 ft.)

The engine was always operated at full throttle and braked with a Ranzi hydraulic dynamometer. The ignition advance was timed in each case so as to obtain the maximum torque.

We used a Zenith carburetor suitably modified to enable a wide range of the mixture ratio. The fuel used was ordinary Shell benzine having a specific gravity of 0.742 at 30°C (86°F).

The inducted air was measured by an orifice plate suitably calibrated by the Montel method and connected with a sensitive alcohol micromanometer (See Gramber, Technische Messungen, Edi-

tion VI, p. 193). In order for the orifice plate to operate under favorable conditions, it was located at about 2/3 of the length of a pipe 115 mm (4.5 in.) in diameter and 1.5 m (nearly 6 ft.) long joined to the pipe of 35 mm (1.38 in.) diameter. The previous theoretical investigations indicated that this would not affect the inertia tests.

Figure 11 shows the arrangement of the apparatus, comprising the hydraulic brake F with its tank S_1 and the tachometer T; the cooling-water tank S_2 ; the engine M and thermometer t; the fuel tank S_3 with the automatically refilled container S_4 for determining the hourly consumption; and the micromanometer p with orifice plates before and behind the air inlet or nozzle b, and carburetor c.

The formula for determining, from the pressure difference P (in millimeters of water), the quantity of air Q (in kilograms per second), passing through the opening, is

$$Q = \alpha F \sqrt{2 g \gamma P},$$

where F represents the cross-sectional area (in m^2) of the restricted portion of the nozzle b, γ the specific gravity of the air, and α a coefficient to be determined by calibration.

In order to obtain this coefficient, the additional pipe is connected directly to a centrifugal blower, the quantity of air flowing through the pipe being regulated by varying the speed of the blower. All the tests gave the same coefficient $\alpha = 1.004$. However, while the air current generated by the

blower can be kept perfectly constant, the current generated by the engine is periodically very variable and may even be reversed for brief periods as shown by curve b in Figure 9.

The manometer connected with the air inlet does not indicate the mean value but that which, in the theory of alternate magnitudes, is called the "effective value," i.e., the square root of the mean square. If the air current varies, the coefficient α must also involve the ratio between the mean value and the effective value. This ratio depends on the shape of the curves of the air velocities. It would be 0.75 according to the calculations used as the basis of Figures 3 and 4. It was determined experimentally by resorting to a capacity of about 70 liters (2.47 cu.ft.) (S_5 of Fig. 11), which could be put in parallel, between the pipe containing the inlet nozzle b and the additional pipe, by means of a valve f. The theoretical researches, already mentioned in Section VII, gave assurance that the parallel arrangement of the capacity would satisfactorily serve the double purpose of steadying the velocity behind the nozzle b and in the additional pipe, and of rendering the flow through the measuring orifice almost perfectly uniform.

For the interpretation of the results of the tests made without the capacity in parallel, it was finally decided to adopt 0.6 as the value of the coefficient α .*

*See footnote next page.

*These tests were made before the nozzle b was calibrated. After this was done and the excessive valuation resulting from the application of the coefficient 1.004 was thus confirmed, we recognized the necessity of adding the air capacity in parallel.

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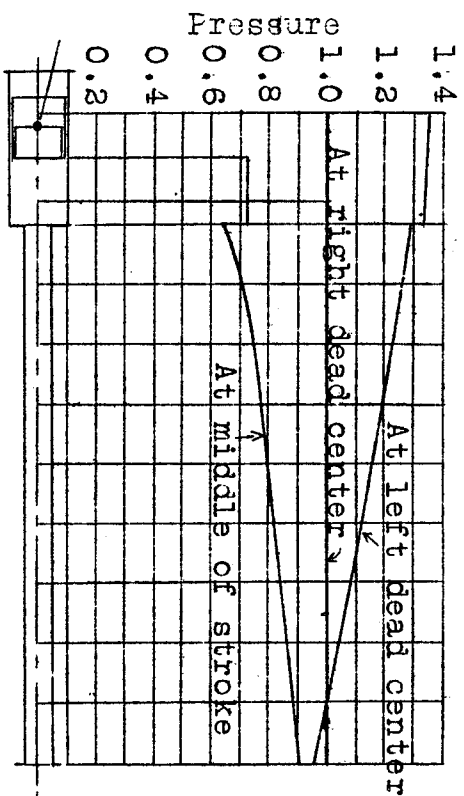


Fig. 1

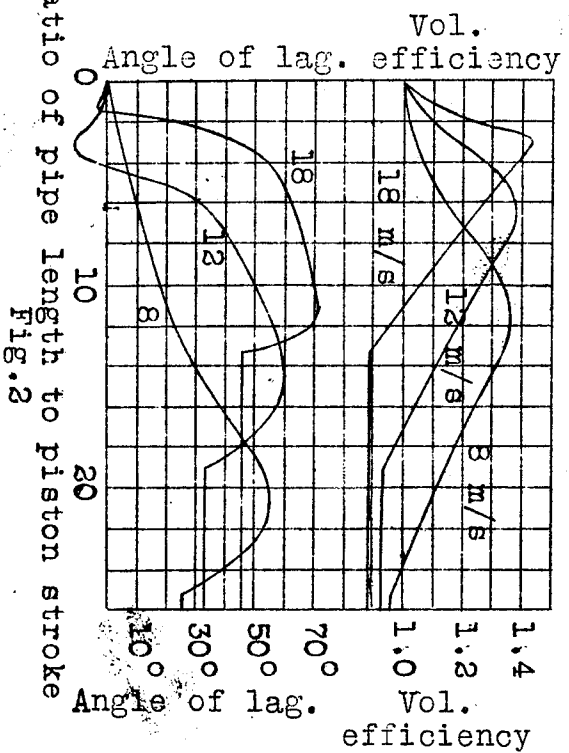


Fig. 2

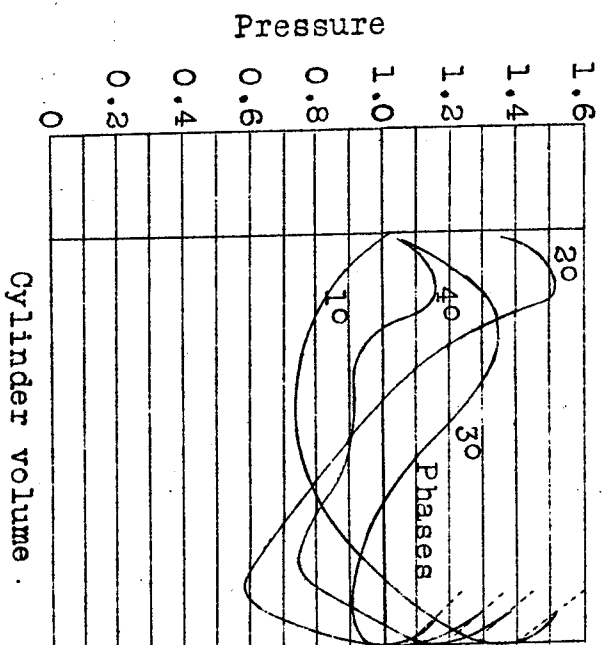


Fig. 3

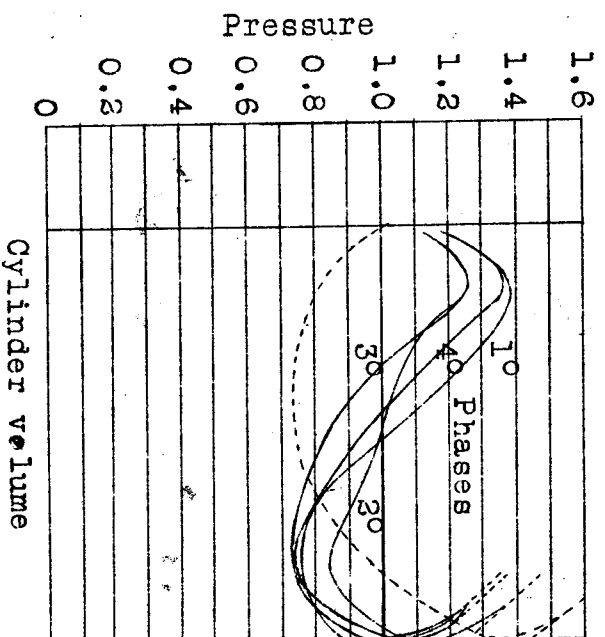


Fig. 4

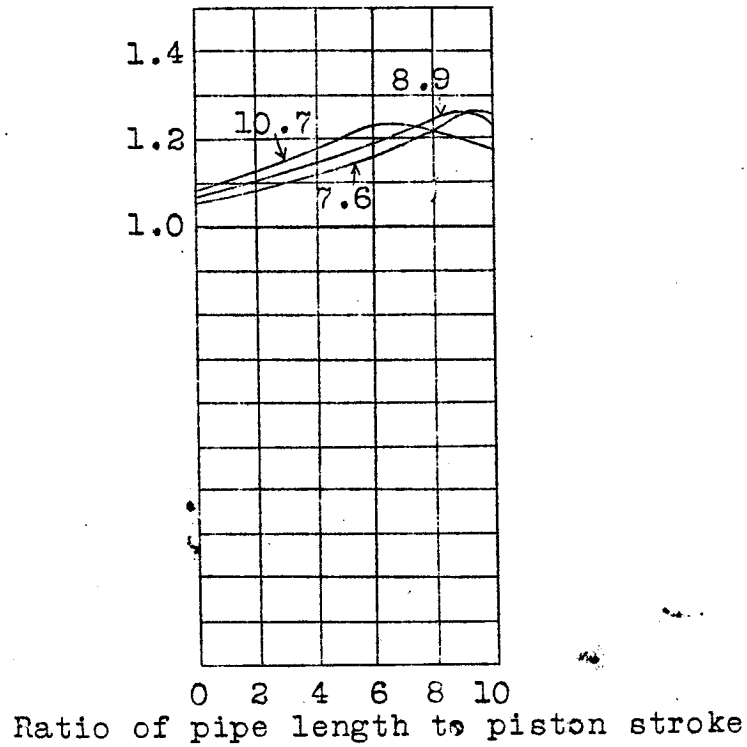


Fig.5

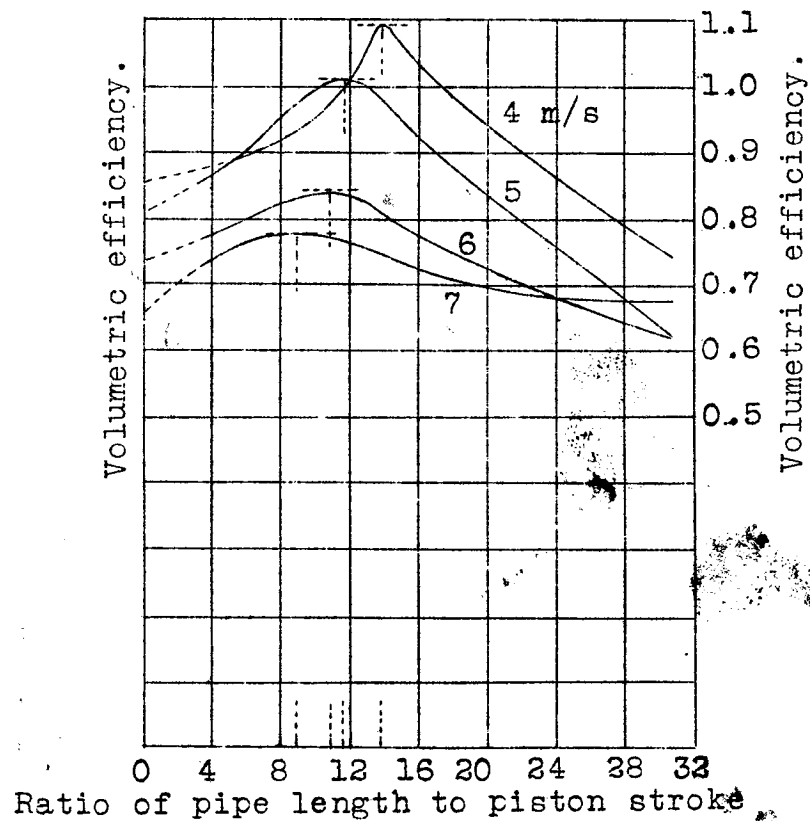


Fig.6

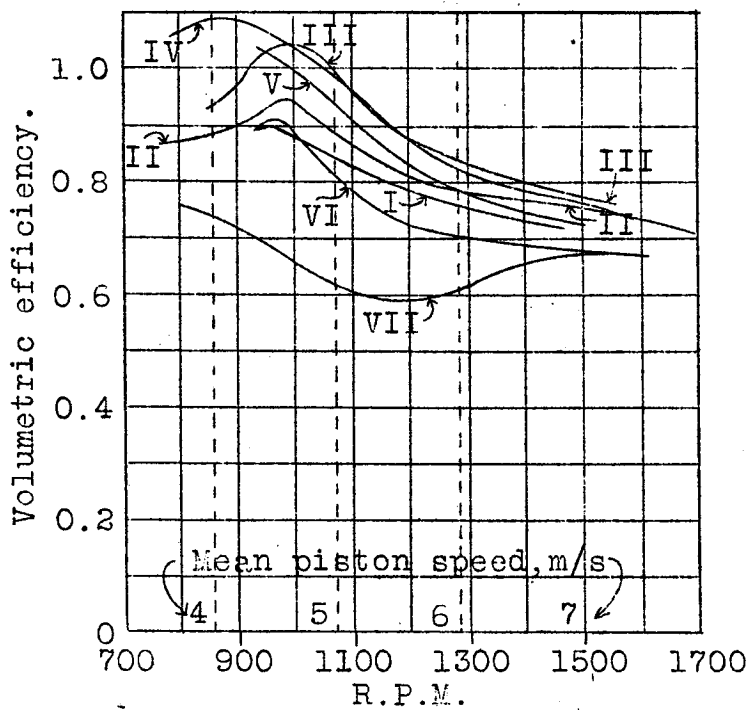


Fig. 7

Curve

Length of additional pipe.

I	0
II	0.35
III	0.93
IV	1.48
V	1.62
VI	2.60
VII	3.92

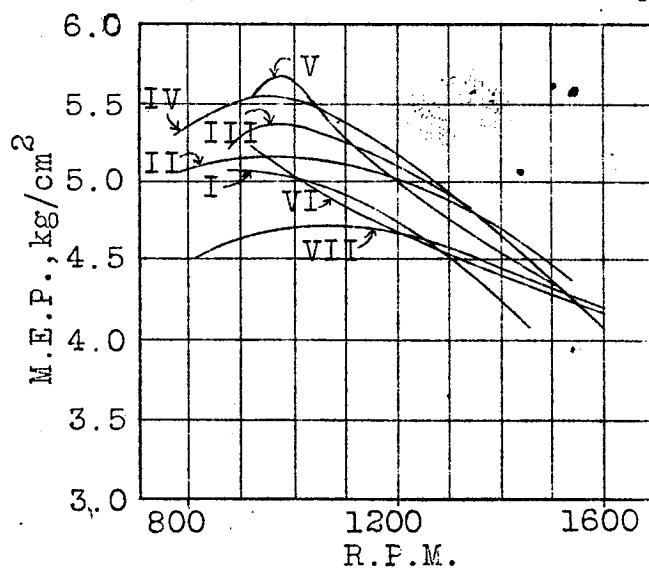
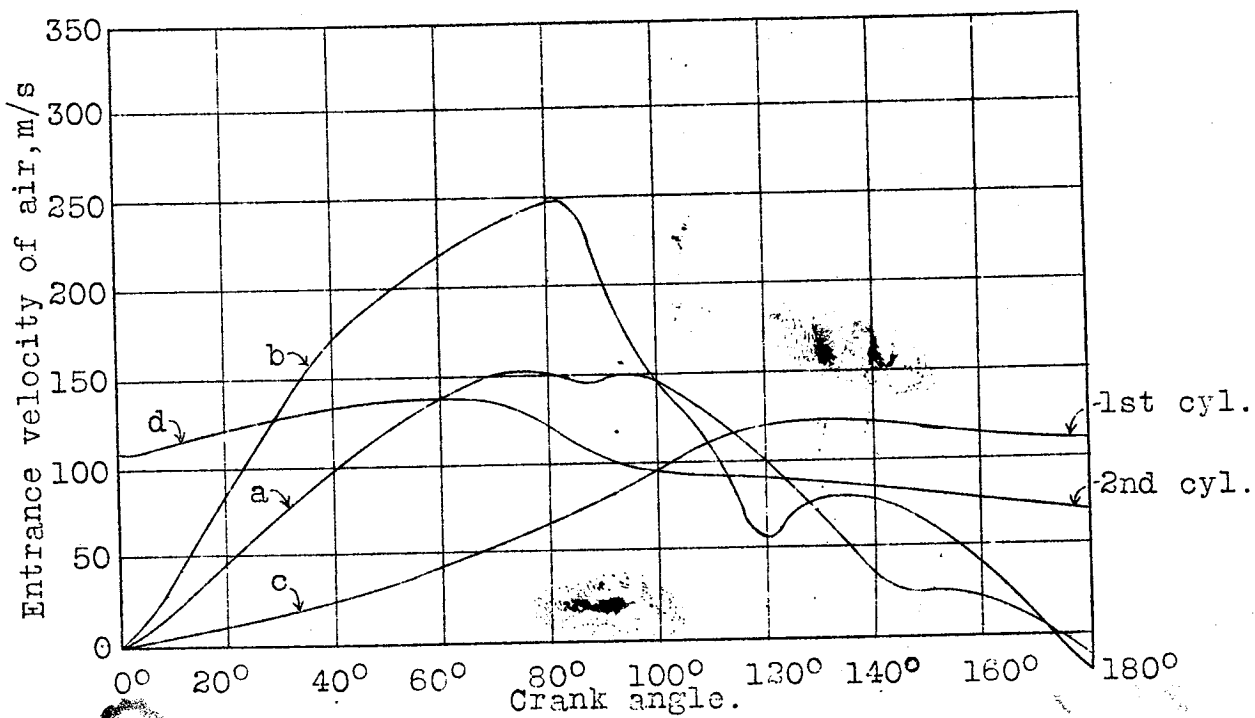
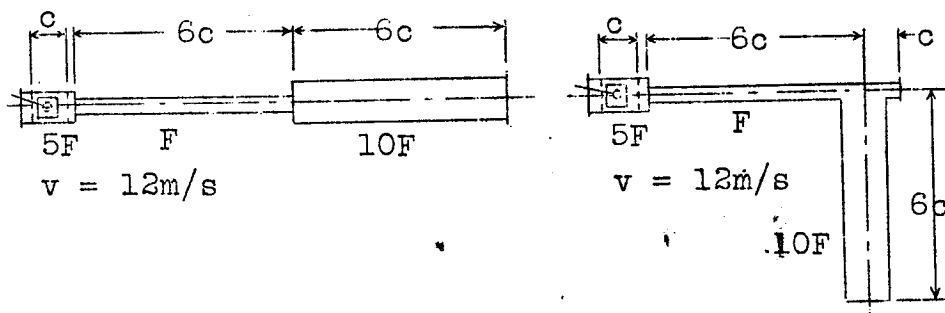


Fig. 8



a, Without tank.

b, With tank in series.

c, With tank in parallel.

d, With tank in parallel.

Fig. 9

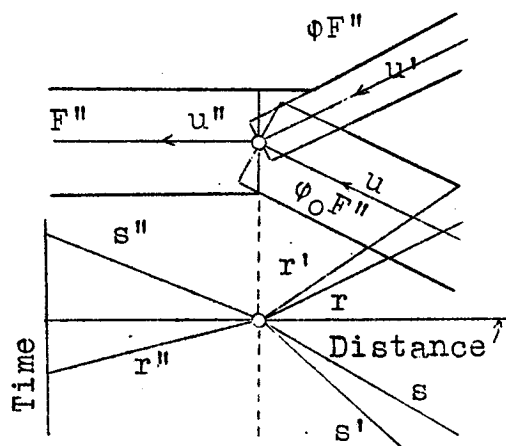


Fig.10

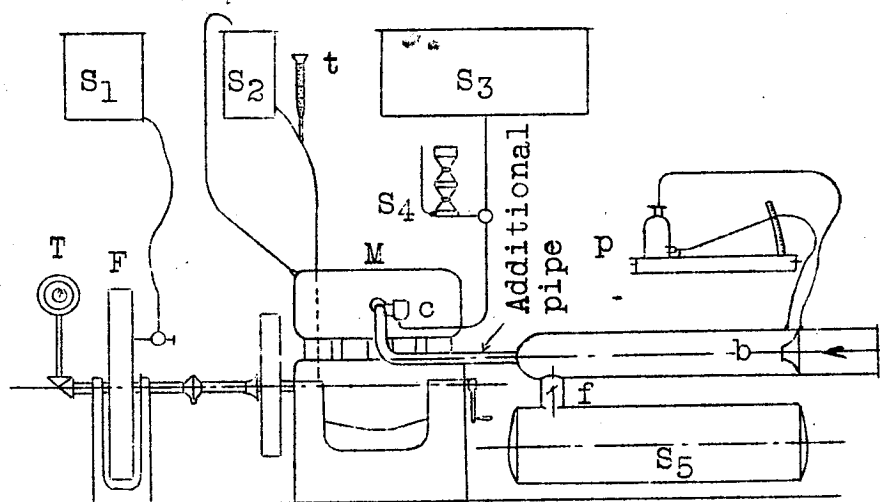


Fig.11

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